THE RUCKER COMPANY (A)

A Centrifuge Project

In September, 1964, Mr. Vincent Salemme, then recently hired by The Rucker Company, Oakland, California, was faced with the problem of designing a centrifuge. The company had been awarded a contract by the Picatinny Army Arsenal in Dover, New Jersey. The device was to have a capability of operation with 4', 6' or 12' centrifuge arms, and was to be so constructed that the intermediate size arm could be accelerated to 232 rpm in 100 milliseconds with an 80 pound load at the end of the arm. (This is termed the high onset mode.) The smallest arm was to have a 10 pound load and be brought to 1240 rpm in 4 seconds, while the largest was to carry a load of 125 pounds to a speed of 213 rpm in 8 seconds. The device was to be finished in six months and was to be complete with a hydraulic system (for motor and control power) and all necessary control and monitor electronics.

Rucker was at this time a company of 225 people, 80 of whom were occupied with the engineering, design, and research aspects of the business. They were engaged primarily in research, engineering, and manufacturing. A specialty was automatic control systems for commercial, marine, aerospace, and defense applications. These products were assembled mostly from purchased parts. Rucker had previously built a centrifuge with a 50 ft. arm for NASA. The company had in 1964 net sales of almost 11 million dollars per year.

Prepared in the 1965 Summer Design Workshop at the University of California, Berkeley, by Øivind Lorentsen and Allen Krauter under the direction of H. O. Fuchs with support from the National Science Foundation. Revised and reproduced in the Engineering Case Program at Stanford University, Stanford, California, 1965.

Exhibit 1: Excerpts from TSL Specification No 40 "Facility, Test, Vibration and Acceleration, Electro-Hydraulic Controlled" issued 13 January 1964 by Picatinny Arsenal, Dover, New Jersey (and revision).

1.0 Scope

The intent of this specification is to procure a single integrated system which will provide rotational and linear test capabilities as outlined. The system shall include a three mode centrifuge and accessories, a low frequency vibrator unit, a common power supply, and a control and monitor center. The control and monitor center shall contain controls for each of the above units with programming and monitoring functions. The system shall be furnished as a complete factor-tested unit; shall be installed at Picatinny Arsenal, Bldg. 3109; and made ready for operation.

2.1.1 Basic Rotary Motor:

(a) The basic servo controlled motor shall have the following characteristics:

Torque:	16,000 inch pounds minimum
Pressure:	3,000 psi maximum
Speed:	1,200 rpm nominal

(b) The spindle bearings shall be heavy enough so that the output shaft can directly sustain a continuous 3500 pound radial load without damage; externally lubricated bearings will not be acceptable.

(c) Control of this motor will be achieved by a two-stage hydraulic valve.

2.1.2 High Onset Drive

(a) High onset torques shall be obtained on a six (6) foot arm. 6000 g pounds shall be available. The arm shall accelerate to 55 g (min) in 100 milliseconds (max).

3.1.2 Programming - Automatic Control

3.1.2.1 - Centrifuge

(a) Acceleration or velocity program shall be obtained through use of an arbitrary function generator, or a programmer. The generator shall be capable of producing a non-linear command from a ramp input. The non-linear command shall be bounded by 10 segments with slope adjustment to 0 to ± 10 volts/volt change per segment based on ± 100 volts output. The adjustment between adjacent break points shall be 0 to 40 volts referred to the input, with parabolic rounding adjustment of curvature from break points of 0 to ± 30 volts. The sweep rate of this generator shall be adjustable from 10 msec. to 100 seconds, and be either single sweep or repetitive.

(b) A programmer shall be supplied which will produce a non-linear command with time from a curve drawn on 11 inch by 17 inch graph paper. The sweep rate shall be adjustable from 0.5 to 50 seconds per inch. The accuracy of this non-linear command shall be ± 1% of full scale for any program limited to 20% full scale change/second.

3.3.6 The centrifuge shall be capable of:

3.3.6.1 - Rise from 0.001 g to 55 g in 100 milliseconds with a package weight of 80 pounds and 1 sq. foot area (the total aerodynamic drag area on each side must be at least one square foot), with a radius of gyration of 36" holding 55 g continuously, and returning to 1 g within one second. Arm length 6 ft. minimum; the arm diameter is approximately 6 feet 4 inches. Maximum bare table "g" shall be 120 g.

3.3.6.2 - The period of time to decelerate the 80 pound package weight to zero speed shall be 200 millisec. after a one second run at 55 g. No accumulators are required to achieve this deceleration.

3.3.6.3 - Rise from zero g to 100 g in eight seconds, with a package weight of 125 pounds and 2 sq. foot area, with a radius of gyration of 78", holding 100 g continuously, and returning to zero g within 12 seconds. Arm length 12 ft. minimum. Arm diameter shall be approximately 13 feet 4 inches. Maximum bare table "g" shall be 175 g.

3.3.6.4 - Rise from zero g to 1000 g in 4 seconds with a package weight of 10 pounds and 0.5 sq. foot area, with a radius of gyration of 23" holding 1000 g continuously, and returning to zero g within 6 seconds. Minimum table length shall be 4 feet. Arm diameter shall be approximately 4 feet 3 inches. An arm is acceptable for the specified 4 foot diameter wheel.

3.3.6.5 - The inertia of the drive system, less arm, shall not exceed 0.5 in lb. sec^2. The centrifuge drive system shall be designed to accept other arms and controls to permit onset rates higher than those specified

3.6.2 For acceptance, the equipment shall be given a continuous operational time duration test. This shall be performed at Picatinny Arsenal after the final installation checkout. The total time of this test shall be at least 24 hours and consist of test durations of at least 6 hours each. The test items shall be dead weights, (maximums), contractor supplied. Operations shall include single and/or dual operational testing. (centrifuge/vibrator).

3.7.1 Six (6) complete instruction and maintenance manuals shall be supplied. The manuals shall include a complete listing of all component parts and assemblies.

3.7.2 The contractor shall provide complete instructions to Picatinny Arsenal personnel in the calculations, operational techniques, and maintenance of this facility.

THE RUCKER COMPANY (B)

Inertia Starter for the Centrifuge

Mr. Salemme was faced with the decision of what type of stored energy to employ to accelerate the six foot centrifuge arm in the high onset mode. While considering alternatives, he was constrained by the need for fast action and low cost, by the customer's specifications, and by the necessity for a compact, integrated system.

The stored energy approaches evaluated were springs, compressed air and mechanical kinetic energy. The first two he eliminated because of both their complexity relative to the chosen method and their higher cost. Various devices to couple these types of stored energy to the centrifuge were also brought up, one of these being a spool and cable mechanism (as used to start lawn mower engines). He finally settled on kinetic energy in the form of a flywheel and listed simplicity and cost as primary reasons for his decision.

The flywheel choice required a decision as to the connection of this rather large mass to the motor, the shaft, and the horizontal centrifuge arm. Before making a choice, Mr. Salemme made calculations of the moment of inertia of the flywheel based on an assumed maximum flywheel speed of 800 rpm. He also calculated the torque necessary to accelerate the intermediate size arm. With these figures he could evaluate the general requirements for the device to couple the flywheel rapidly to the arm. He then attempted to determine a system configuration and a method of rapid momentum transfer so that the cost and time involved in design and fabrication would be minimized.

Mr. Salemme had to find a means of rapidly coupling the flywheel to the arm. He briefly considered fluid couplings, but rejected them as being too complex, permitting slip to take place, and necessitating a higher initial flywheel speed. The simplest device he could envision was a clutch, and he settled on that concept. A dry clutch was selected due to its simplicity and the fact that the coefficient of friction was higher than in a wet clutch.

Mr. Salemme checked with manufacturers of complete clutch units as well as those who used clutches. He soon discovered that no item could be bought off the shelf ready for his use.

He first considered mounting the flywheel and clutch below the motor and the centrifuge arm with all four elements on a common vertical axis. By mounting the clutch above the flywheel in this manner, he would be able to switch easily from the high onset mode to normal operation of the centrifuge by simply leaving the clutch disengaged. In order to keep the system inertia low, he felt it necessary to mount the motor with its axis the same as that of the centrifuge. This required a double shaft motor, and he could find none. He then deemed it necessary to go to another method of designing the clutch - flywheel - centrifuge assembly that would keep the inertia low but would employ standard parts. Also of prime importance was minimum height for the whole design.

Another concept evaluated by Mr. Salemme was that of a flywheel with a horizontal axis; however, he saw that the resulting device would not be compact and that some means would have to be found to "turn" the angular velocity since the arm had to rotate about a vertical axis. The added complication would result in increased system inertia.

<u>Suggested Questions</u>

1. How would you arrange and couple the centrifuge arm hydraulic motor(s), clutch and flywheel about a vertical axis (or axes)?

2. Is there an optimum flywheel speed? If so, how would you determine it? If not, explain why not.

3. Do you think that Mr. Salemme was premature in deciding upon a dry clutch at this stage? What would you want to calculate before choosing or designing a clutch? How?

THE RUCKER COMPANY (C)

Centrifuge Clutch Design

The flywheel - clutch - arm configuration that Mr. Salemme finally evolved is illustrated in Exhibit 1. In normal operation, the clutch is left disengaged, and the main hydraulic drive motor is used alone. For high onset operation (with intermediate size arm), the sequence of events is as follows:

The small motor, A in Exhibit 2, is activated, which causes the flywheel B to turn, driven through the clutch plate lifters K which ride in slots in the outer clutch drum. Upon its reaching speed, pressure is applied to the hydraulic fluid in space C where it causes parts D and E to move downward. The shoe E presses the clutch plates together. Bearings G allow part E to turn while allowing D to move only vertically. As the clutch plates L and M or N are alternately fixed to the flywheel B (through splines in the drum S) and the shaft R, the latter begins to turn. Simultaneously, the main motor F (Exhibit 1) is activated, which helps in bringing the shaft (and arm) up to speed. The main motor is a Vickers size 2125 piston type motor displacing 6.1 in^3/rad. It is driven by a Denison Series 62 variable displacement pump.

Upon reaching speed, part D is raised by applying pressure to space I. The spring H separates the plates by raising the lifter K when the hydraulic pressure is removed.

This design upon which Mr. Salemme decided was built around a crawler tractor steering clutch, the clutch which engages the tread. In looking through the specifications for clutch materials, Mr. Salemme noticed a reference to current usage in earth moving equipment. This led to the Caterpillar Tractor Company. As used by that company in their D8 model tractor and shown in Exhibit 3, Mr. Salemme felt the device to be too complicated. Hence, only inner and outer drums J and S, Caterpillar part numbers 6B2049 and 2B1579 (drum S not shown in Exhibit 3) were purchased from Caterpillar. The clutch plates used in the D8 clutch are standard Velvetouch items made by the S.K. Wellman Division of the American Brake Shoe Company. A drawing of the internally toothed disc appears in Exhibit 4; the externally toothed disc (M and N) is shown in Exhibit 5. The main ingredient in the sintered material which is bonded to the O.D. Gear Tooth Disc is copper.

Mr. Salemme modified half of the externally toothed discs by cutting three slots in them (plates N in Exhibit 2). The detail of this modification is shown in Exhibit 6. The slotted plates are alternated in the stack. This allows every second plate (plates M) to be lifted by spring H and lifter K during disengagement, the slots providing clearance for the lifter past the

nine discs N which are not lifted. The lifter raises the upper plates more than the lower plates.

The height of parts J and K and the thicknesses, number, and general dimensions of the clutch plates were fixed. Specifying the flywheel to have the same height, he made the radius such that it had the required inertia about its axis.

Mr. Salemme obtained the friction coefficient for the materials from the manufacturer. He made calculations to determine what normal force was required to obtain the necessary torque. He checked this with the clutch plate manufacturer and was told that he was within the acceptable range. He did not calculate the clutch life because the number of times the high onset capability was to be used did not exceed several hundred per year. Also, he felt that Caterpillar designed for hard, continuous usage, and that despite their use of the device as a wet clutch, it would still be well overdesigned in the centrifuge application.

No heat transfer calculations were made; Mr. Salemme felt this not to be a problem in the design.

One notices that the flywheel is not completely separated from the shaft even when the clutch is not engaged; only every other plate is disengaged by the spring lifter. Mr. Salemme felt this simplified the design. Since the normal force between each rubbing surface is only a few pounds when the clutch is disengaged, he believed the heat generated posed no problem. No brake was provided on the flywheel; it is allowed to come up to speed even when the high onset mode is not being used. Mr. Salemme felt this to be desirable in that it minimized steady state clutch wear. He noted that the flywheel motor could be turned off to prevent flywheel rotation should this prove necessary.

Mr. Salemme decided it would be cheaper to make the flywheel in layers than in one solid piece. This is illustrated in Exhibit 2. He checked the flywheel bearing with the manufacturer; it was well beneath any expected loadings. The two bearings G were specified to insure stability of Part E as it moved vertically to engage the clutch. He said, "They are just loafing along in this application." He noted that all the bearings in the design were oversize and that they were strong enough to withstand the loading produced by the test piece falling off at speed.

All materials used in the design were chosen by Mr. Salemme on the basis of his experience. No formal optimizations were made either of dimensions or materials. He felt that the time was too short and that the goal was to produce a working piece of equipment in the least time and at minimum cost.

Mr. Salemme estimated that the clutch and flywheel cost about $2800 including the material and engineering work. This was as expected. The total cost of the engineering and mechanical hardware of the centrifuge he felt to be about $20,000. This did not include the motors,

hydraulics, or electronics. He explained that the flywheel - clutch work involved about 56 hours of his engineering time; a draftsman contributed an additional 40 hours of layout work and 40 hours of drafting time.

Upon assembly, the machine was turned by hand to check the clearances. An interference was seen to exist. The machine was disassembled; part O was found to rub against part Q where the latter was welded P. The weld was made smaller. Subsequent assembly revealed no further interferences.

On May 16, 1965, tests were initiated to determine whether the customer specifications had been met. Mr. Salemme noted that full testing was not possible at the California location since no adequate test cells were available. Those tests which were completed were done without the load, remaining test results being predicted by Mr. Salemme's analysis. All the tests proved satisfactory; the clutch worked as planned except for an excessive velocity overshoot in the high onset mode. He believed this to be a problem of the control system rather than a difficulty with the clutch itself.

Certain other difficulties were revealed by the testing. According to Mr. Salemme, the center of shaft rotation was 0.006 inches from the geometric shaft center. Also noticed was shaft twisting during the high onset phase of operation. Mr. Salemme stated that the solution to the problem would involve a complete redesign of the centrifuge: the flywheel would have to be moved closer to the arm itself. He stated that if he had to start the design over, he would do this.

At present, no checks have been made to determine the condition of the clutch or bearings. Before shipment to the customer, the machine is to be disassembled and the more critical parts inspected. Mr. Salemme did not anticipate any trouble and was optimistic that the solutions to the above problems would soon be found.

<u>Suggested Questions</u>

1. Would you choose to support the flywheel on a single large-diameter bearing?
2. Suggest alternative ways of driving the flywheel.
3. Suggest alternative ways of keeping the plates separated when the clutch is disengaged.
4. Calculate the torsional deflection of the shaft during high onset acceleration. How would you solve this problem in a redesign?
5. Estimate the lowest torsional frequency of the centrifuge.

GENERAL CENTRIFUGE LAYOUT

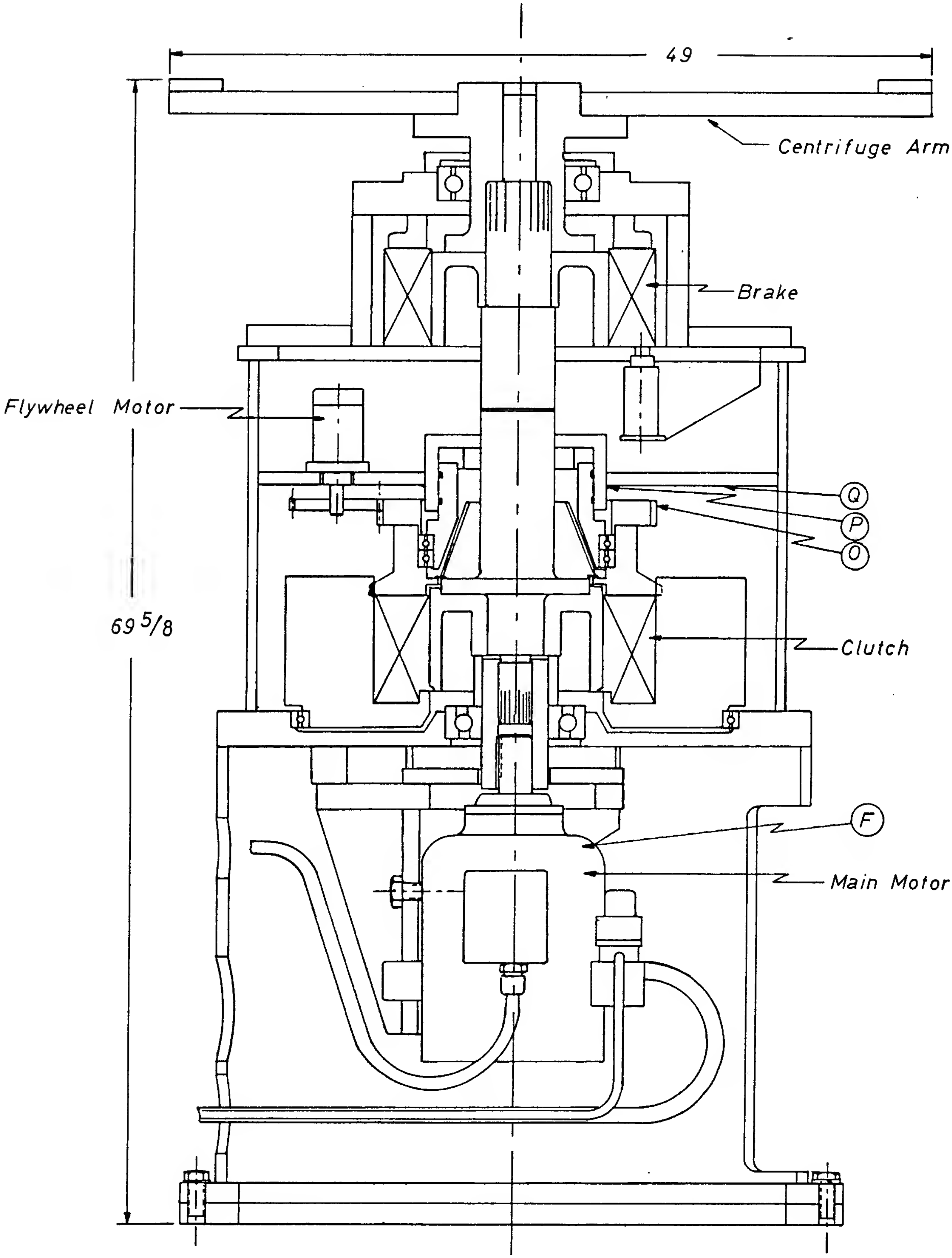
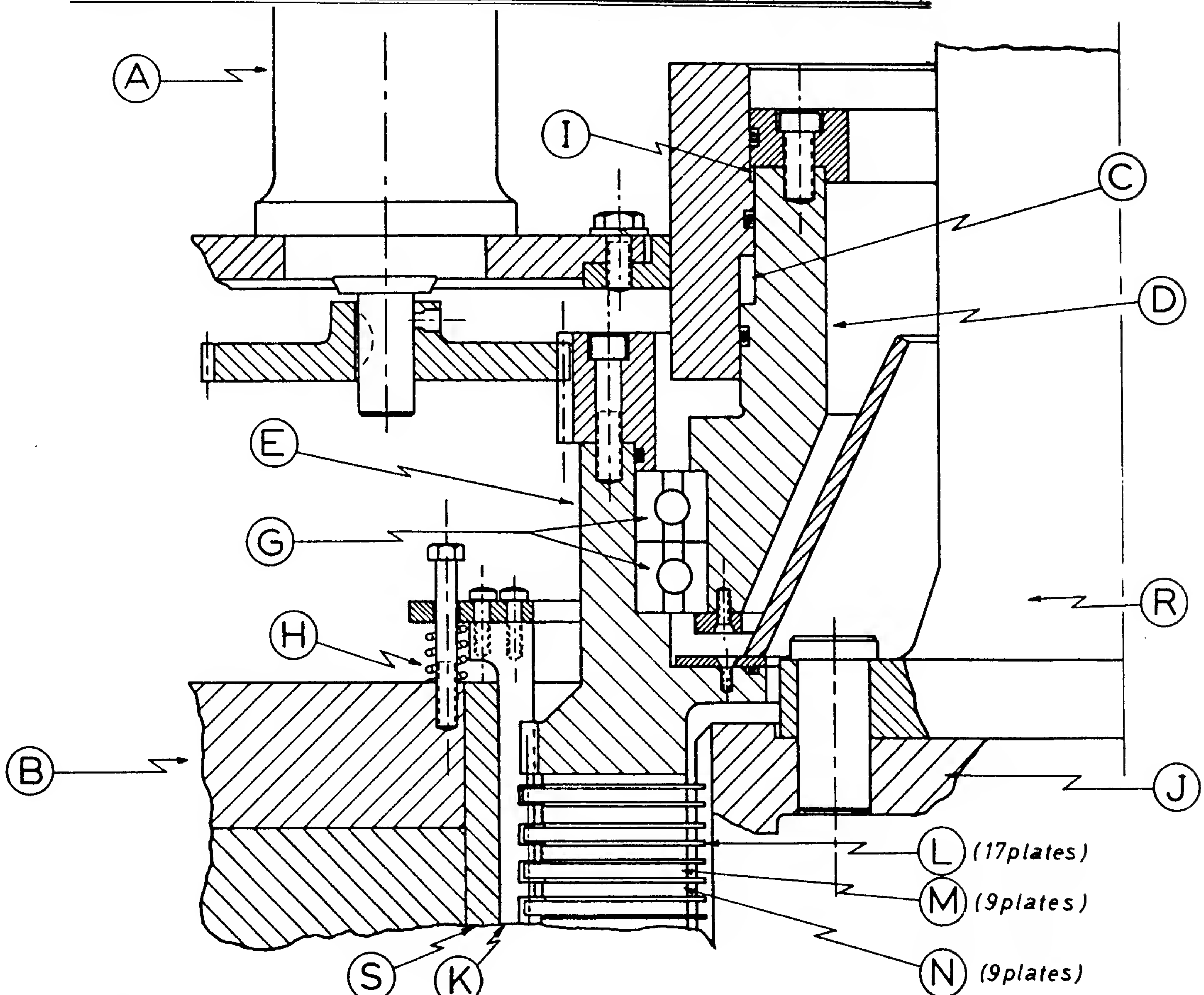


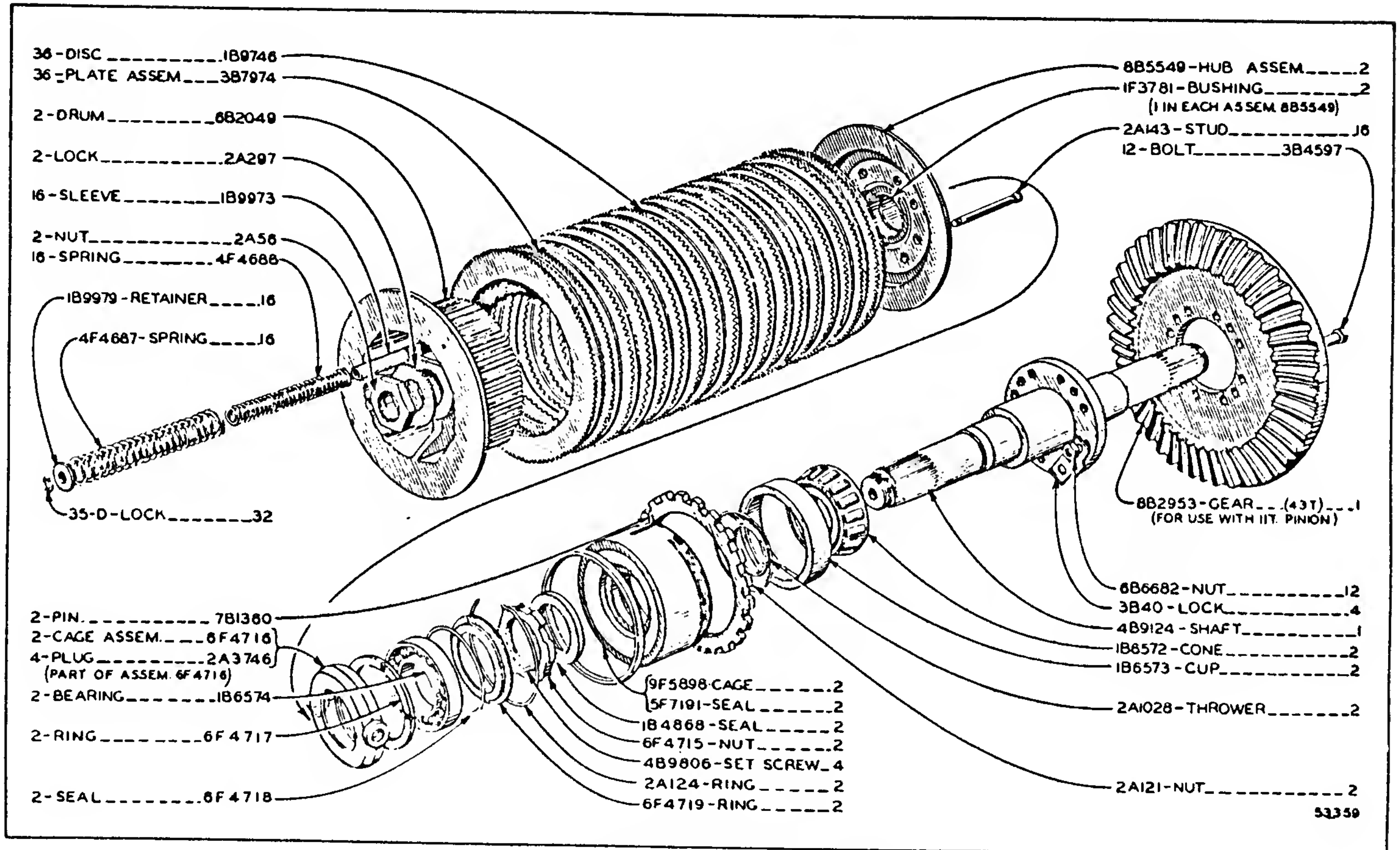
Exhibit 1: Mechanical Elements of the Centrifuge

DETAIL OF CLUTCH FLYWHEEL SYSTEM



SCALE: 2"

Exhibit 2: Flywheel Drive Mechanism and Clutch Actuation Details



STEERING CLUTCH
Serial No. 2U16900 to 2U21512 inclusive

Exhibit 3: Steering Clutch for Caterpillar Model D8 Track Type Tractor

ECL 39

Exhibit 4

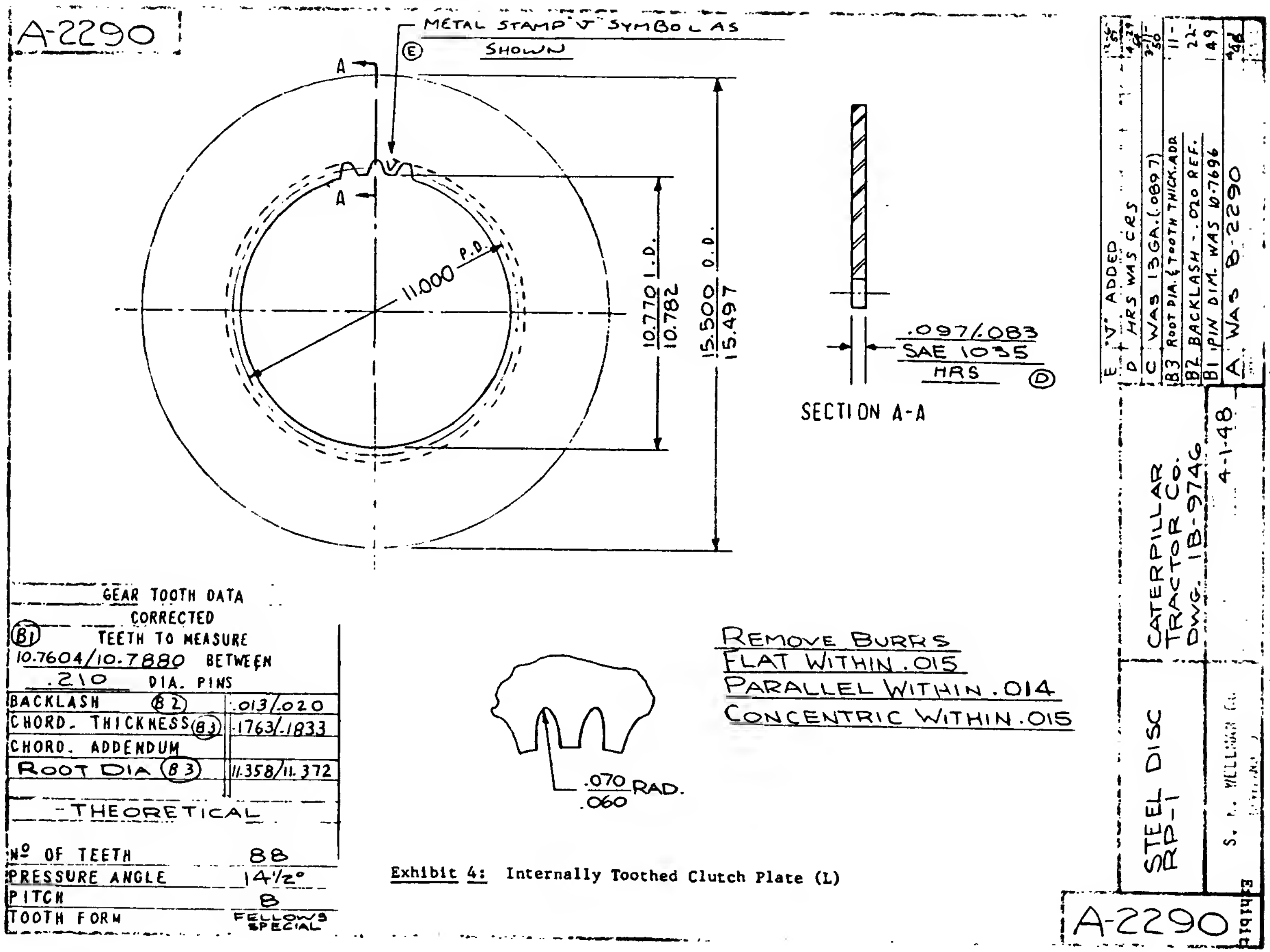


Exhibit 4: Internally Toothed Clutch Plate (L)

ECL 39

Exhibit 5

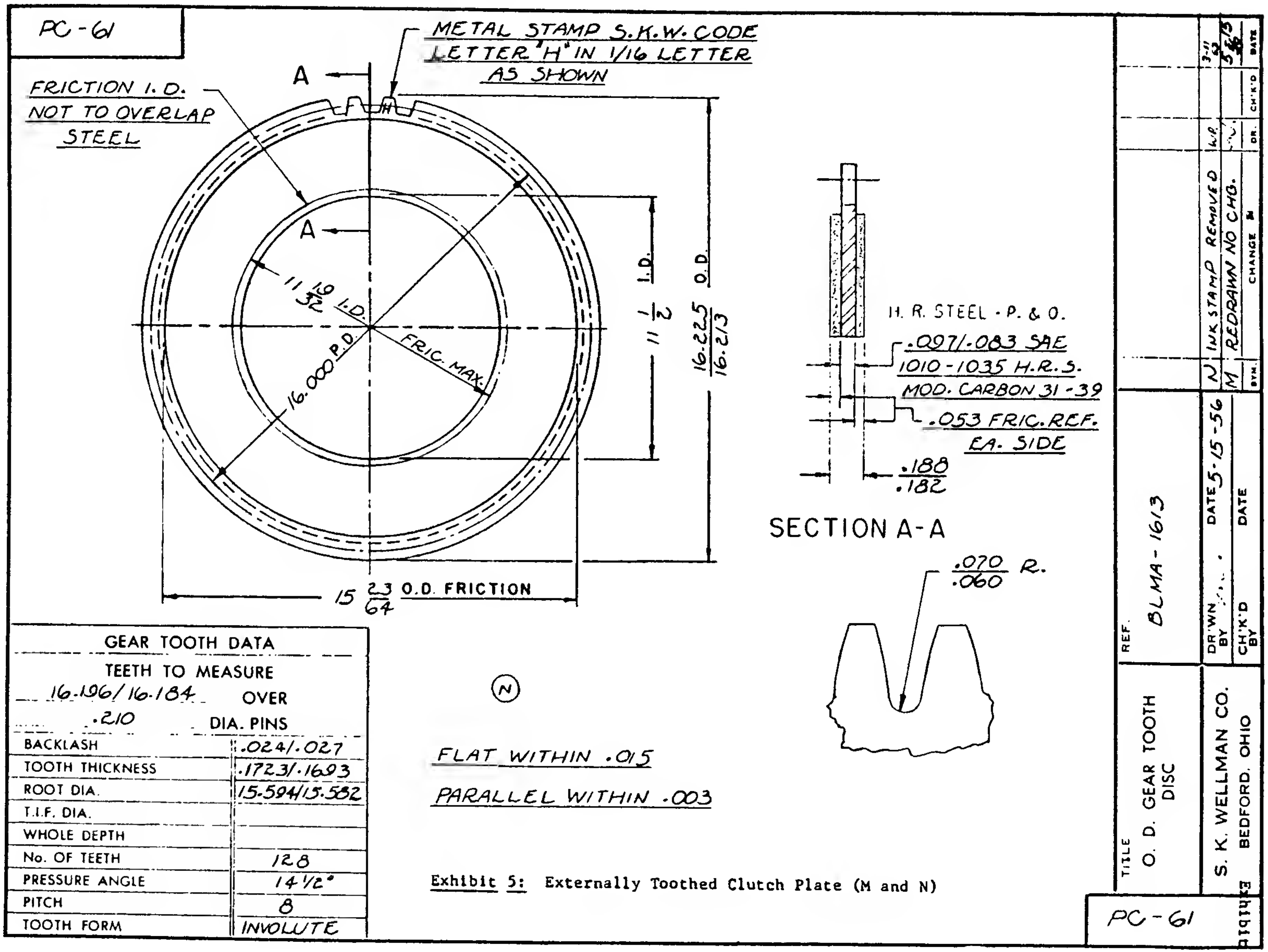


Exhibit 5: Externally Toothed Clutch Plate (M and N)

Friction Disc

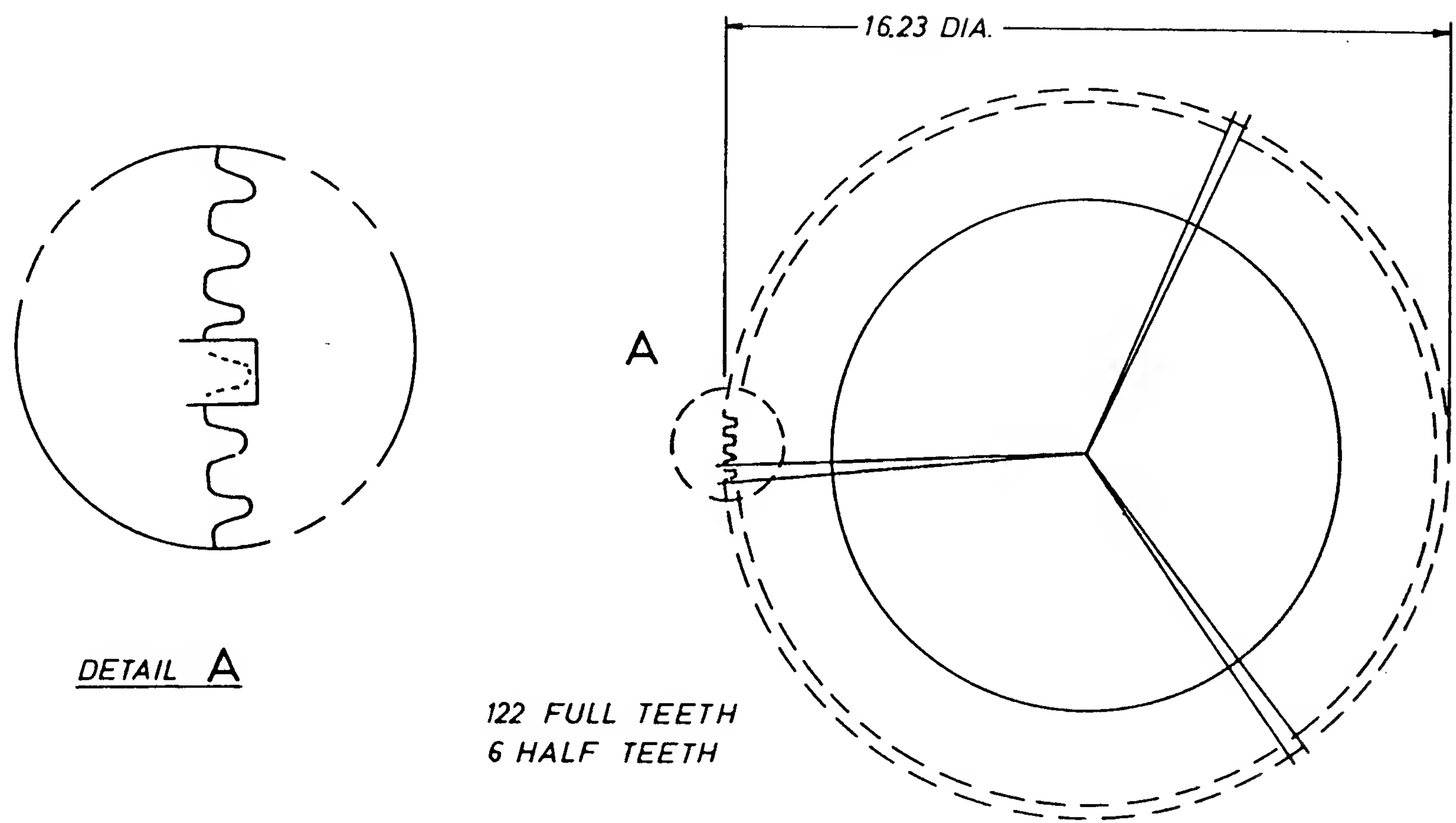


Exhibit 6: Modified O.D. Gear Tooth Disc with Slots Providing Clearance for Plate Lifters (Alternate Discs Only)

In the spring of 1964, Rucker had presented to the Arsenal a bid for the centrifuge and a vibrator, both powered by the same hydraulic system. Their price was about 1/4 million dollars (including installation), which happened to be about $1000 below that of the nearest competitor. The Rucker management envisioned a stored energy approach to deal with the high acceleration requirements of the centrifuge. The high onset specification was new to both Rucker and to centrifuge design in general.

Mr. Salemme, a mechanical engineering graduate of the University of Alabama (1943), was given the centrifuge design as his first problem. "I just happened to walk through the door," he said of his hiring. He had experience in electrical products and control applications. He had been employed by Western Electric, where he had been engaged in the design of machinery for telephone cable manufacturing. Mr. Salemme was not sure that he had the required depth of experience in hydraulics when he was first named Project Manager on the centrifuge project.

The stored energy method had been decided upon by management just a day before the bid deadline. At the outset of the project, Mr. Salemme might have proposed a different method of achieving the rapid acceleration, but he felt that the proposed method was workable, saying he couldn't see another way to do it and that time would not permit much reflection. He had considered a motor capable of the required output, but noticed that the case with the high onset of speed required more torque than did the other cases. It would therefore not be practical to design the prime mover for that one situation. He also considered a gearing device for multiplying the torque from a motor of sufficient horse power. However, he found that the system inertia would then be greater than the limit set by the customer (See Exhibit 1).

Picatinny Arsenal had specified certain requirements to be met by the centrifuge because of the nature of their activities. They were engaged in work dealing with high explosives and would use the centrifuge in this application -- for example, testing projectile fuses. The Arsenal therefore stipulated that the centrifuge motor be hydraulic rather than electric to minimize the danger of explosions and also that the centrifuge be able to operate safely should the test specimen separate from the arm. Exhibit 1 consists of excerpts from the specifications given Rucker.

<u>Suggested Questions</u>

1. How would you store the energy needed to accelerate the centrifuge in the high onset mode?
2. Sketch the plan for the first week of Mr. Salemme's work on this project.